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Project director(s):
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Sponsor/division names: SKY CLIMBER INC / ST MOUNTAIN, GA
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Title: POWER HOIST CONCEPTS

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Security class (U,C,S,TS) : U ONR resident rep. is ACO (Y/N): N
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PROJECT CLOSEOUT - NOTICE

17-SEP-1997 16:37

Closeout Notice Date 17-SEP-1997

Project Number D-48-X43

Doch Id 46928

Center Number 10/24-6-R8850-0A0

Project Director HARDY, T.

Project Unit DEAN ARCH

Sponsor SKY CLIMBER INCORPORATED/STONE MOUNTAIN, GA

Division Id 12194

Contract Number AGR DTD 960103

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Title POWER HOIST CONCEPTS

Effective Completion Date 24-MAY-1996 (Performance) 24-MAY-1996 (Reports)

Closeout Action:	Y/N	Date Submitted
Final Invoice or Copy of Final Invoice	Y	26-AUG-1996
Final Report of Inventions and/or Subcontracts	Y	
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Classified Material Certificate	N	
Release and Assignment	N	
Other	N	

Comments

Distribution Required:	
Project Director/Principal Investigator	Y
Research Administrative Network	Y
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Research Security Department	N
Reports Coordinator	Y
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Georgia Tech Research Corporation	Y
Project File	Y

NOTE: Final Patent Questionnaire sent to PDPI

Main Project – D-48-X43
#1

FINAL REPORT
POWER HOIST CONCEPTS

Project A-5161

22 May 1996

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1.0 INTRODUCTION

Sky Climber, Inc., desires to develop and market a new generation of power hoists. They have sought Georgia Tech's assistance in this product re-design. As part of Georgia Tech's multi-disciplinary program with Sky Climber, the Georgia Tech Research Institute (GTRI) has focused on the mechanical design that is needed to develop this next-generation power hoist.

During this program, GTRI reviewed Sky Climber's list of desired capabilities, existing designs (both Sky Climber's and their competitors), and the regulatory environment. A variety of concepts for the key mechanical components (the traction mechanism and drive train) have been generated and evaluated. This report summarizes these findings and other pertinent hoist-related concerns.

These findings can be used by Sky Climber to select a design concept for the next-generation power hoist. The effort required for a detailed design would need to be completed in a follow-on program.

2.0 REVIEW OF SKY CLIMBER'S DESIRED CAPABILITIES

Sky Climber set forth several guiding principles for GTRI to use while proposing new concepts for a power hoist. These principles were as follows:

- Modularity of components;
- Durability (ability to operate in severe environments);
- Reliability (reduce sensitivity of unit to wire rope condition and reduce contaminant build-up);
- Ease of installation (ease of handling, reduced hoist weight, and self-reeving with optional breech loading);
- Ease of maintenance (reduction in the number of hoist components, accessibility to interior components);
- Wire rope handling (accommodate several wire rope sizes and reduce likelihood of wire rope damage during operation);
- Total hoist target weight of 75-100 pounds; and
- Target cost of $1,000.

Sky Climber set forth other guiding principles and requirements at the beginning of the program. These principles are not listed as they were not applicable to the mechanical design during this initial phase of the program. Most of the requirements that Sky Climber set forth were related to the regulatory environment that must be adhered to. Applicable regulatory references were as follows:

- Underwriters' Laboratories (UL 1323);
- American National Standards Institute (ANSI A10.8, ANSI A39.1A, ANSI A120.1);
- Occupational, Safety and Health Administration (CFR 1926.451, CFR 1910.66); and
- European Standard (EN: TC98 WG7).

3.0 REVIEW OF EXISTING DESIGNS

Three power hoists were examined (Sky Climber's Alpha 1000 and Model 4 and Power Climber's Pocket Climber). The information gained was used to assist GTRI in better understanding what is required to develop a power hoist that would meet Sky Climber's desired capabilities.

3.1 Sky Climber's Alpha 1000

The Alpha 1000 hoist, weighing 130 pounds, has a 1,000 pound rated load capacity. The unit utilizes a planetary gear reduction drive (117:1 ratio) and a traction sheave. The wire rope is self-reeved from the inlet guide into the sheave. Pressure is applied to the wire rope through the use of Belleville washers. The wire rope is captured within the sheave for 270° of rotation before it is forced to exit the hoist. The unit accepts only 5/16-inch diameter wire rope.

Common complaints include jamming of the wire rope and wear on the exit piece. Because the traction mechanism and wire rope are enclosed within a housing, the unit is susceptible to contaminant build-up. The exit piece located at the tail end of the wire rope is susceptible to wear.

3.2 Sky Climber's Model 4

The Model 4 hoist, weighing 95 pounds, has a 750 pound rated load capacity. The unit utilizes a worm gear (5:1 ratio) as its primary gear reduction drive. A belt drive provides greater reduction. A traction drum with 4+ wraps of the rope achieves the necessary traction. The drum itself has urethane coated circular grooves. Fairlead guides move the wire rope from one circular groove to the adjacent groove. The unit accepts only 5/16-inch diameter wire rope and can only be breech loaded. The fairlead guides and the traction drum are both susceptible to wear.

3.3 Power Climber's Pocket Climber

The Pocket Climber hoist, weighing 106 pounds, has a 1,000 pound rated load capacity. The unit utilizes a double reduction gear drive with a worm gear as the primary stage. A traction sheave is used as the traction mechanism. The wire rope is captured by the traction sheave for 270+° of rotation before it is forced to exit the hoist. The tail rope is forced against the sheave by a pinch roller. A guide roller guides the wire rope to its exit from the sheave. It is believed that a V-groove sheave is utilized to ensure that the necessary traction is achieved. It is GTRI's understanding that the unit accepts only 5/16-inch diameter wire rope. The hoist can be either self-reeved or breech loaded, although the breech loading process is rather difficult.

GTRI is not aware of any wear problems. It is speculated that the guide roller may be susceptible to wear.

4.0 REGULATORY AND ENGINEERING REQUIREMENTS

Currently, Sky Climber manufactures four power hoists to accommodate lift capacities of 750, 1,000, and 1,500 pounds. There is little, if any, modularity between the different power hoists. Sky Climber desires that their next-generation power hoists have modularity of components.

Sky Climber would like for the new hoist to accommodate the four common wire rope diameters (5/16-inch and 3/8-inch in the United States, and 8mm and 9mm in Europe). To satisfy their desire for modularity, critical components, such as the traction mechanism, must be designed to accommodate the largest wire rope diameter (3/8-inch). This impacts the traction mechanism size, the output torque required, and the output speed (RPM) of the hoist.

4.1 Regulatory Requirements

A plethora of regulations are applicable to the overall design of a power hoist. The following regulations guided the development of the traction mechanism-power hoist concepts.

ANSI A120.1 14.8.2 (Drums & Sheaves) states that traction drums or sheaves shall have a thread diameter 25 times greater than the wire rope diameter. If the next-generation hoist is designed to accommodate a wire rope up to 3/8-inch diameter, the diameter of the traction drum or sheave must be 9.375 inches.

ANSI A.120.1 14.10.4 (Minimum Lift Capacity) states that hoist motors shall lift 125 percent of the rated load. UL 1323 39.1 (Performance) states that the hoist shall ascend or descend while carrying 125 percent of its rated working load. For a 1,000 pound rated load, the hoist must be capable of lifting 1,250 pounds. For a 1,500 pound rated load, the hoist must be capable of lifting 1,875 pounds. These loads were used in determining the operating parameters of the hoists.

UL 1323 5.1 (General) states that the maximum rated speed at which the suspended scaffold may be moved in a vertical direction shall not exceed 35 feet per minute (fpm). This sets forth the maximum lifting speed of the hoist and dictates the output speed (RPM) of the hoist. From this, the required gear reduction ratio is computed.

UL 1323 41.1 (Strength Test) states that a hoist, while suspended, shall be loaded for 5 minutes to 4 times its working load. This requirement sets forth survival loads for the hoist components. For a hoist with a 1,000 pound lift capacity, the components must withstand a 4,000 pound load. Similarly, a hoist with a 1,500 pound lift capacity must withstand a 6,000 pound load.

4.2 Engineering Requirements

The above regulatory requirements were used to determine the necessary engineering parameters. In determining such parameters as torque, horsepower, output speed, and the gear ratio, motor efficiency and gear drive losses were not taken into consideration. Gear drive losses and other parameters would have to be taken into account in the detailed design of the hoist.

Table 4-1 shows the hoist parameters for a 9.375-inch traction drum used in hoists with 1,000 pound and 1,500 pound lift capacities. A 1,750 RPM motor is assumed.

Table 4-1. Traction Drum Driven by a 1,750 RPM Motor

	1,000-lb. Rated Load	1,500-lb Rated Load
Wire Rope Diameter	0.375 in.	0.375 in.
Traction Drum Diameter	9.375 in.	9.375 in.
Traction Drum Radius	4.688 in.	4.688 in.
125% of Rated Load	1,250 lbs.	1,875 lbs.
Output Torque	5,860 in-lbs.	8,790 in-lbs.
Lifting Speed (Maximum)	35 fpm	35 fpm
Horsepower	1.32 hp	2.00 hp
Output Speed	14.26 RPM	14.26 RPM
Input Speed (Motor)	1,750 RPM	1,750 RPM
Gear Reduction Ratio	122.7	122.7
Input Torque (Motor)	48 in-lbs.	72 in-lbs.

Table 4-2 shows the same hoist parameters for a 9.375-inch traction drum used in hoists with 1,000 pound and 1,500 pound lift capacities. In this case, a 3,600 RPM motor is assumed to be the driving motor.

Table 4-2. Traction Drum Driven by a 3,600 RPM Motor

	1,000-lb. Rated Load	1,500-lb Rated Load
Wire Rope Diameter	0.375 in.	0.375 in.
Traction Drum Diameter	9.375 in.	9.375 in.
Traction Drum Radius	4.688 in.	4.688 in.
125% of Rated Load	1,250 lbs.	1,875 lbs.
Output Torque	5,860 in-lbs.	8,790 in-lbs.
Lifting Speed (Maximum)	35 fpm	35 fpm
Horsepower	1.32 hp	2.00 hp
Output Speed	14.26 RPM	14.26 RPM
Input Speed (Motor)	3,600 RPM	3,600 RPM
Gear Reduction Ratio	252	252
Input Torque (Motor)	23 in-lbs.	35 in-lbs.

It can be seen that a hoist with a 1,000 pound lifting capacity requires a 1.32 hp motor, while a hoist with a 1,500 pound lifting capacity requires a 2.0 hp motor. If the higher speed motor is utilized, the gear reduction ratio is increased, but the input torque is decreased. There could possibly be some weight savings in going with the higher speed motor. This is discussed in detail in Section 8.0.

One traction mechanism concept that is being proposed (see Section 7.0) is a chain traction drive. In this concept, 3.622-inch diameter sprockets are used to drive two roller chains which apply traction to the wire rope. The smaller diameter sprocket reduces the output torque and the required gear reduction ratio. Table 4-3 shows the engineering parameters if a 1,750 RPM motor is assumed. The reduction in the gear ratio is significant.

Table 4-3. Sprocket Driven by a 1,750 RPM Motor

	1,000-lb. Rated Load	1,500-lb Rated Load
Wire Rope Diameter	0.375 in.	0.375 in.
Sprocket Diameter	3.622 in.	3.622 in.
Sprocket Radius	1.811 in.	1.811 in.
125% of Rated Load	1,250 lbs.	1,875 lbs.
Output Torque	2,264 in-lbs.	3,396 in-lbs.
Lifting Speed (Maximum)	35 fpm	35 fpm
Horsepower	1.32 hp	2.00 hp
Output Speed	36.9 RPM	36.9 RPM
Input Speed (Motor)	1,750 RPM	1,750 RPM
Gear Reduction Ratio	47.4	47.4
Input Torque (Motor)	48 in-lbs.	72 in-lbs.

5.0 GEAR DRIVE CONCEPTS

Various gear train types were explored for the next-generation power hoist. These included a compound gear train (spur or bevel gears), a worm gear drive, a planetary (epicyclic) gear drive, and a harmonic drive.

Of these gear trains, the worm gear drive and the planetary gear drive would be best suited. Both gear trains can be utilized to achieve high gear reduction ratios in a small envelope.

5.1 Worm Gear Drive

Worm gears provide the easiest way of greatly decreasing rotation speeds between two shafts. Unfortunately, there are several potential problems with worm gears that the designer must account for.

Worm gear drives are not as efficient as most other gear trains and often have a hard time getting above 90 percent efficiency. Friction losses can be high as the worm threads slide sideways along the worm gear teeth.

Worm gear drives are susceptible to self-locking (drive cannot be reversed) if the reduction ratio is too high or the helix angle is too low. Worm gear drives are also susceptible to "stairstepping" when operated with an overhauling load. Stairstepping is an erratic rotation of the gearset. This erratic rotation can be amplified by the rest of the gear drive creating a very undesirable operating condition.

The potential problems of self-locking and stairstepping can be prevented by using worm gear ratios of 15:1 or less. Lead angles greater than 11° will allow the worm gear drive to be backdriven. Multiple worm threads can be used over single threads to increase the drive efficiency.

As shown in Section 4.0, calculations have shown that a 122:1 gear reduction ratio would be required if a 1,750 RPM motor were utilized. A high ratio like this would necessitate the use of a multiple reduction reducer (double reduction worm gear drive or a helical/worm gear drive).

5.2 Planetary Gear Drive

A planetary gear configuration is typically used to achieve a high gear reduction ratio in a small envelope. Planetary gear drives share loads between several meshes thus leading to significant envelope and weight savings. The smaller, stiffer components associated with a planetary configuration drive lead to reduced noise and vibration. The load-carrying capacity of the planetary can also be increased by increasing the number of planets. Planetary gear arrangements are also highly efficient (97 - 99 percent).

A wide variety of gear reduction ratios can be achieved with a planetary gear drive by varying the member sizes (sun, ring, or planet gears), adding an additional set of planet gears, or by varying which member is fixed and which is linked to the output shaft.

5.3 Gear Drive Selection

Worm gear drives and planetary gear drives are both utilized on existing power hoists. Hence, either configuration can be made to work with Sky Climber's new generation hoist.

A planetary reducer can provide high efficiency in a light, compact package. Hoist envelope restrictions may make a right angle drive desirable, leading to the selection of a drive combining worm and parallel shaft gearing.

6.0 DUAL DRUM TRACTION HOIST CONCEPT

6.1 Cable Path

The cable path of the dual drum hoist is shown in Figures 6-1, 6-2, and 6-3. The cable wraps around two grooved drums, mounted one above the other with horizontal axes of rotation. The axes are twisted approximately 2.2° in opposite directions about a vertical axis. This twist aligns the grooves of the two drums so the fleet angle is approximately zero. There is no need to separate the drums to minimize the angle, so a compact hoist is feasible.

The number of wraps is chosen to provide adequate traction. This presentation shows three full wraps or six half wraps.

6.2 Traction

The traction capability of the dual drum hoist depends upon the number of wraps, groove geometry, coefficient of friction, and pinch roller forces. As indicated in Section 4.1, the operating requirement for a hoist with a 1,500 pound lift capacity is 1,875 pounds. The static test requirement is 6,000 pounds.

For this analysis, the coefficient of friction, μ, is assumed to be 0.15. The V-groove angle affects the apparent coefficient of friction, f.

$$f = \mu / \sin\theta$$

where θ is half the angle between the sides of the groove. The smaller the angle, the harder the rope wedges between the sides. This improves traction, but also increases crushing loads on the rope (this issue will be addressed in more detail below). In addition, small angles may cause the rope to jam in the groove. The condition for onset of jamming is

$$\mu = \tan\theta.$$

A groove half angle of 27.5° has been chosen for this presentation. The effective coefficient of friction is then

$$f = 0.15/\sin 27.5° = 0.325.$$ Additionally,

$$\tan\theta = 0.52 \text{ and } \mu = 0.15,$$

so the rope should not jam.

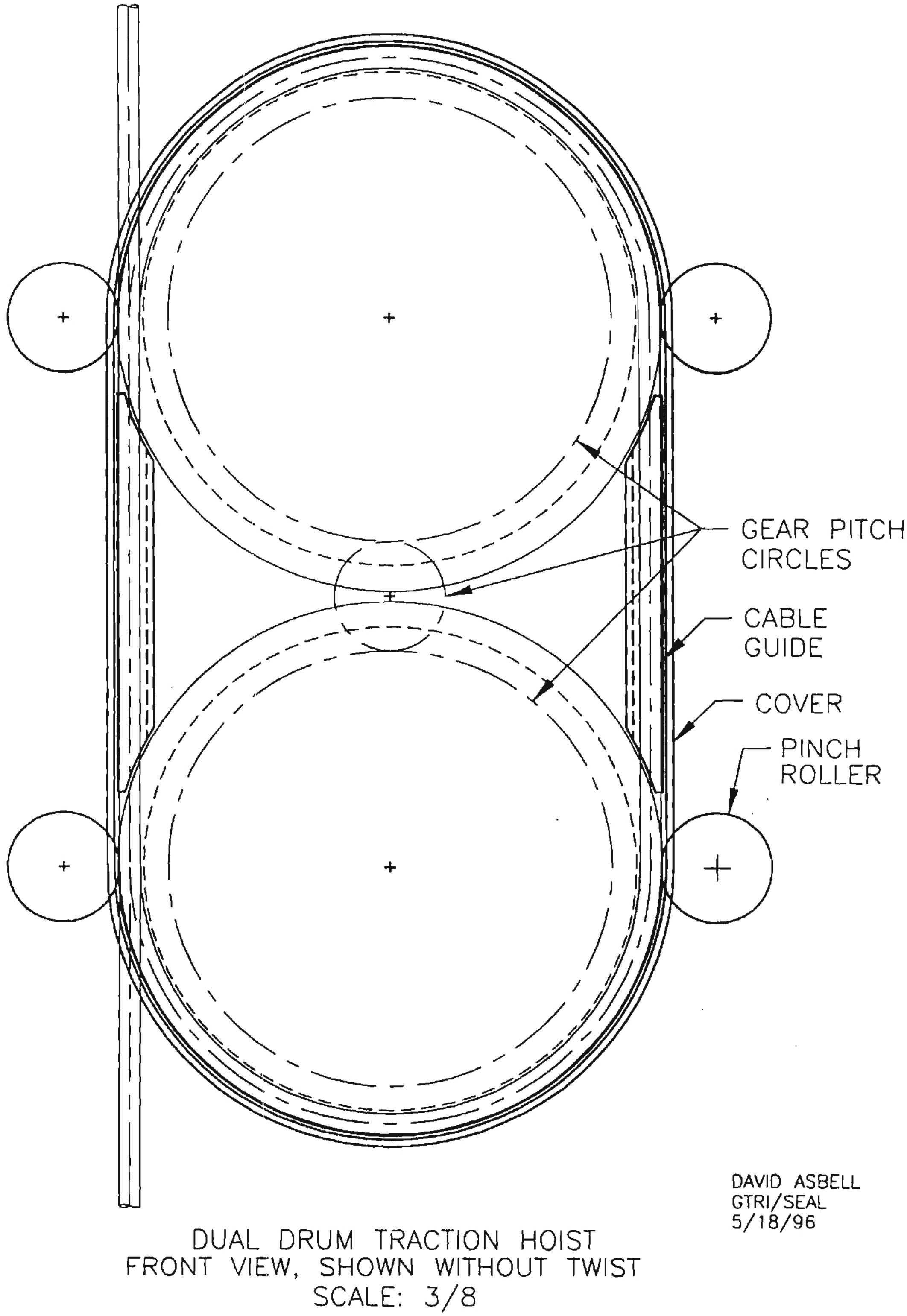

Figure 6-1. Dual Drum Traction Hoist (Front View)

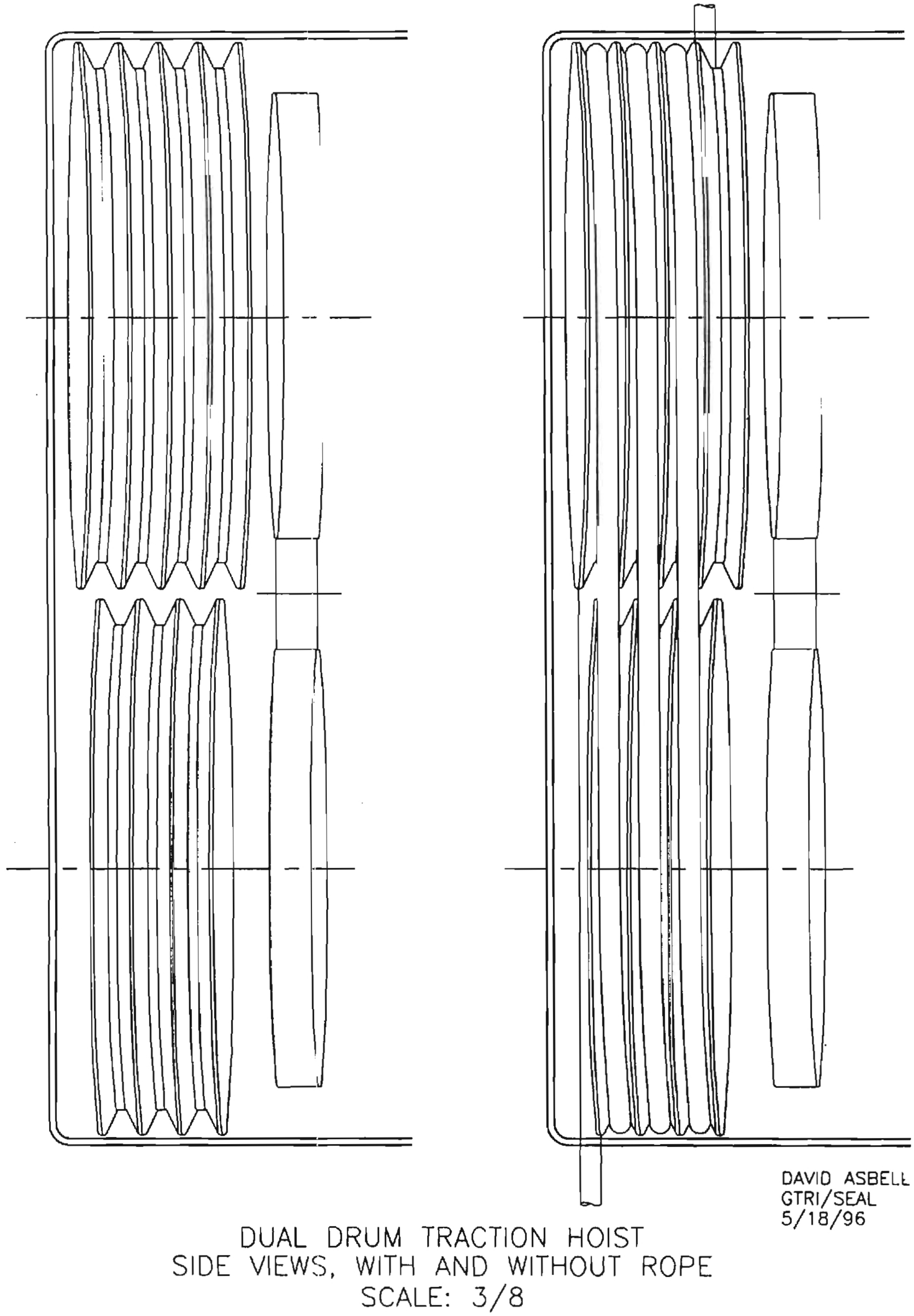

Figure 6-2 . Dual Drum Traction Hoist (Side View)

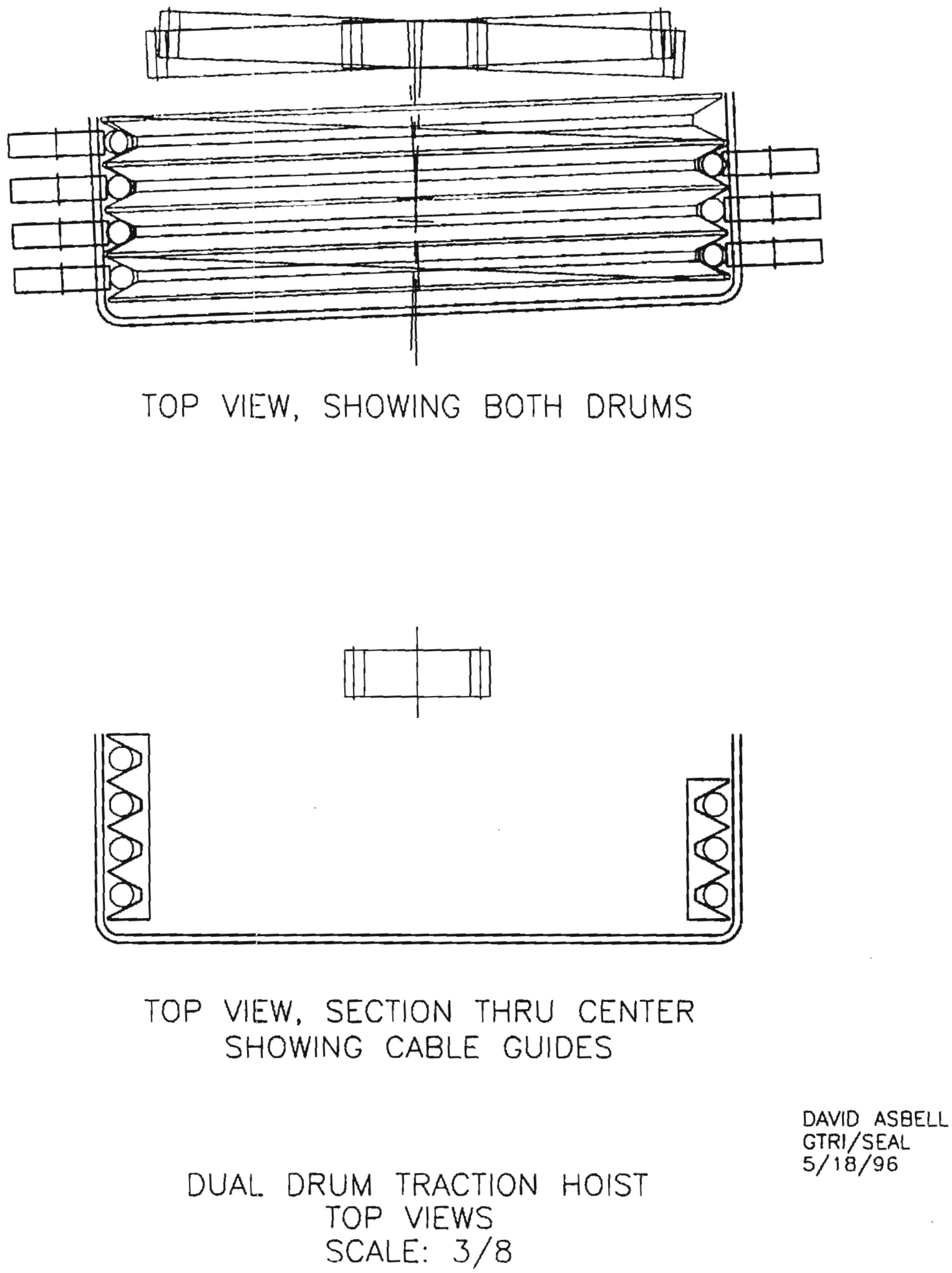

Figure 6-3. Dual Drum Traction Hoist Concept (Top View)

The traction available is

$$T_f = T_i e^{f\phi}, \text{ where}$$

T_f and T_i are final and initial tensions, and
ϕ is the wrap angle in radians.

Setting the final tension to the static test condition of 6,000 pounds, the wrap angle to 6π radians (six half wraps), and f to 0.325, and solving for initial tension, one finds that an initial tension of 13 pounds is required to prevent slip. This condition will be met if a pinch roller applies a force of 13/f, or 40 pounds, to the rope at the point where the tail rope seats on the top drum. This analysis neglects the bending stiffness of the rope which reduces the net seating force. Additional pinch roller loading is needed to compensate for this.

6.3 Rope Kindliness

The dual drum hoist treats the rope very well. The rope does not rub against itself or any stationary object nor does it slide on the drum. It is not subjected to heavy pinch roller loads, and there are no reverse bends. It is forced to bend around the drums and is subjected to compressive loads from the sides of the grooves.

The tension in the rope has a radial component which forces the rope into the groove. This component is

$$F_r = T/R \text{ (pounds/inch), where}$$

T is the rope tension, and
R is the pitch radius of the drum (9.375/2=4.688 inches).

The groove applies compressive forces normal to the walls,

$$F_w = F_r/2\sin\theta \text{ (pounds/inch).}$$

Using the parameters chosen above, one finds

$$F_w = 346 \text{ pounds/inch}$$

for a final tension of 1,500 pounds, where the top rope seats on the bottom drum. This mechanism is inherently load sensitive. In the 6,000 pound static load test, F_w will be 1,386 pounds/inch. If the actual load is lower, the groove forces will be lower.

The equations show that increasing the drum diameter lowers groove forces and bending stresses. However, size and weight constraints suggest using the minimum diameter allowed by the regulations (drum pitch diameter equal to 25 times cable diameter, or 9.375 inches for 0.375 rope).

Increasing the groove angle reduces groove forces. For this very reason, the apparent coefficient of friction f falls. More wraps, or higher pinch roller loads, are then required. GTRI currently has no data relating rope damage to groove forces. The wire rope manufacturer, MacWhite, in Kenosha, Wisconsin, has been contacted.

6.4 Self Reeving and Breech Loading

The dual drum hoist can be designed for both self reeving and breech loading. The keys to self reeving are: spring loaded pinch rollers at each point of rope arrival and departure on the drums; driving both drums; and stationary rope guides. The self reeving process begins with the hoist fully assembled, the pinch rollers spring loaded against the drums, and the hoist running in the up direction. The rope end is inserted into the top rope guide. When it reaches the top drum and roller, it is seized and pushed down the stationary guide chute to the bottom drum. The guide chute is formed by a grooved plate between the drums and the cover. At the end of the chute, the rope is seized by the bottom drum and a roller and, guided by the curved surface of the cover, carried around under the drum. This process is repeated as the rope end reaches each roller, until it exits the tail rope guide. Only the last roller is critical to the traction function. The others may have lower loadings. Clearance between the drums and the curved ends of the cover must be adequate to accommodate the stiff rope tip.

Breech loading the dual drum hoist requires complete access to both drums so the rope can be wrapped into the grooves. Removing the cover provides this access. The key is to design the self reeving features to allow easy removal of the cover. Towards this end, the cover has been made smooth on the inside. The drums have deep grooves, so the rope seats below the outer circumference of the drum. The pinch rollers are mounted on the outside of the cover, on spring loaded toggles so they can be retracted. The twisted axes of the two drums require the cover to be twisted slightly as it is removed. This should present no difficulties. Regulations require that the cover remain attached. A lanyard should achieve compliance without interfering with access.

6.5 Gear Drive

The drums must both be positively driven in the same direction. This requirement is complicated by the fact that the axes are not parallel, but twisted about 4.4° with respect to one another. Low angle helical gearing offers an elegant solution, shown in Figures 6-1, 6-2, and 6-3. A central pinion with a 0° helix angle drives the two drum gears, with left and right hand helix angles of 2.2°. These gears can be cut on ordinary hobbing machines, at no special expense. They function much like straight spur gears, because the helix angles are so low.

The pitch diameters are 2.000 and 8.000 inches. This allows the drums to be mounted 10.000 inches apart, as close as their outer diameters will permit. A reduction of 4:1 is achieved in this final mesh, reducing the required torque capacity of the primary reducer by 75 percent. The pinion has an ample number of teeth, 14, at 7 pitch. No tooth load calculations have been made. The choice of primary reducer type is not specific to the dual drum design, although a right angle gear reducer may have to be utilized to satisfy one of Sky Climber's specifications for the next-generation hoist. This specification states that the hoist must fit through an 18 inch round opening. If the gear reducer and motor were mounted in-line with the central pinion, this requirement would not be satisfied. GTRI needs to consult with Sky Climber regarding this specification and the reason for it, as this specification impacts all of GTRI's concepts.

6.6 Durability, Reliability, and Maintenance

The basic components of the dual drum hoist can be quite durable. Gearing and bearing stress control is straightforward. Contaminants can be effectively excluded, and lubricants retained, by seals and enclosures. Drums and rollers will be subject to wear by the cable, but wear can be limited by material selection and treatment. The drums can be induction hardened, case hardened, or flame sprayed with an abrasion resistant coating. Polyurethane rollers may offer good life. The drums should clear themselves of contaminants, but the cover may retain them and require clearing. The cable will not touch the inside of the cover, or the cable guides, except during the self reeving process. The cable path is smooth and well controlled, and the mechanism is exceedingly simple. A very reliable machine should be the result.

The only frequent maintenance foreseen is clearing contaminants out of the cover when operating in a dirty environment. If the cover is secured with over-center draw latches, and the pinch rollers are mounted on spring loaded toggles, removing and replacing the cover should be very quick. Periodic maintenance would include gear lubrication and wear checks.

7.0 CHAIN TRACTION HOIST CONCEPT

7.1 Cable Path

The wire rope passes straight through the hoist, pressed between a pair of chain loops, as shown in Figure 7-1. Each link of the chains carries a steel V-block which guides and grips the rope. Each chain is stretched between a pair of sprockets. The lower sprockets of each chain loop are driven. Each chain passes around a pressure bar, similar to a chain saw's bar, which guides the chain and presses it against the wire rope.

7.2 Traction

Each chain is pressed against the rope with a bar force, F_b, spread among a number of V-blocks. The traction force available is a function of the force F_b, the coefficient of friction between the V-blocks and the rope, μ, and the half-angle of the V-blocks, θ. An effective coefficient of friction, f, may be calculated as shown in Section 6.2. The traction force available from both chains, T, is

$$T = 2F_bf.$$

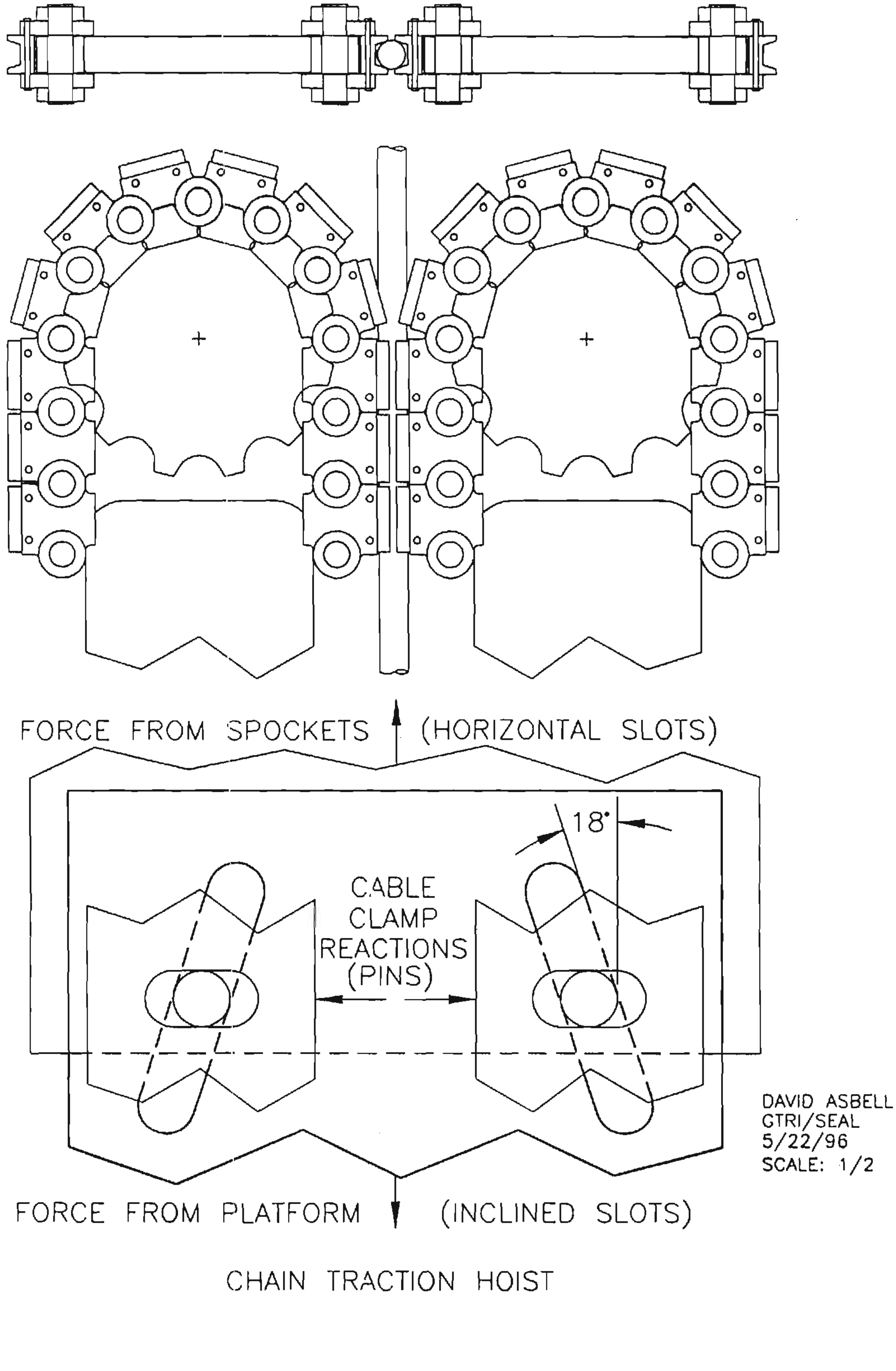

Figure 7-1. Chain Traction Drive Hoist Concept

The chain drive may use a smaller V-block angle than the dual drum, because pressure along the length of the rope is more uniform, and peak loads are lower. For this example, however, a half angle of 27.5° and a coefficient of friction of 0.15 are assumed. The resulting value of f is 0.325. The static test load of 6,000 pounds would then require a bar force F of 9,230 pounds, while the operating load of 1,875 pounds would require a bar force of 2,885 pounds.

The bar force is not inherently load adaptive but needs to be made so. One approach is shown in Figure 7-1. The sprockets are mounted to a structure bearing two horizontal slots. The base of the hoist is attached to a plate with two inclined slots. The pressure bars each have a pin which passes through the crossed horizontal and inclined slots. When a tension load passes through the assembly, the pressure bars are forced together, toward the rope. If the slot inclination angle, α, is 18°, a tension of 6,000 pounds produces bar forces of

$$F_b = T/2\tan\alpha$$
$$F_b = 9{,}233 \text{ pounds, where}$$

F_b is the bar force,
T is the load supported by the hoist, and
α is the slot inclination angle.

The bar force generated by the load adapter is adequate to generate the needed tractive force. An initial spring preload on this mechanism would be needed. The stresses in this mechanism may be quite high where the slots and pins intersect. No analysis has been done.

7.3 Rope Kindliness

The chain traction hoist subjects the rope to no abrasion or bending, only to the compressive loads of the V-blocks. This load is nominally uniform over all active blocks, and the load per block can be adjusted by changing the number of active blocks. For example, the bar load above, 9,233 pounds, spread over an active chain length of 8.44 inches (9 links, with a V-block length of 7.875 inches) would produce a V-block loading of 1,172 pounds/inch. This is comparable to the peak loading of 1,280 pounds/inch for the dual drum hoist. Greater active lengths would give lower loadings per inch. The V-blocks would need generous end chamfers to avoid cutting the rope fibers. This would increase the needed active chain length.

7.4 Self Reeving and Breech Loading

Self reeving requires inserting the top rope tip between the chains while the hoist is operating in the up direction.

Breech loading requires separating the two chains, placing the rope between them, and re-closing the chains. The load adapter mechanism can be utilized to spread the pressure bars. The sprockets have a fixed separation. The cover, serving no function except protection, could be made easy to remove.

7.5 Gear and Sprocket Drive

The two lower sprockets are geared together by a pair of identical spur gears, so they turn at the same speed in opposite directions. One of these gears is driven by the primary reducer. The choice of primary reducer type is not specific to the chain traction hoist. Driving the lower sprockets relieves the upper sprockets of any significant load.

7.6 Chain

The chain is a modified roller chain, with hardened steel V-blocks riveted to the side plates of every link. The side plates are connected by pins. Bushings roll on these pins. The bar load is transmitted from the bar, to the bushings, to the pins, to the side plates, to the V-blocks, and finally to the rope. Each chain carries nominally half the rope tension. The chain tension loads pass from the sprocket teeth, to the bushings, to the pins, to the side plates, to the V-blocks, and finally to the rope. The bushings roll on the pins, and the side plates rotate on the pins, under heavy loads. The result is friction and wear. The amount of friction has not been estimated, but it could cause substantial power loss and heat build-up, with possible lubricant degradation. The wear problem may be severe, given the impossibilities of keeping the chain isolated from contaminants carried by the rope and of providing oil bath lubrication.

The chain is an integral part of the hoist drive mechanism. Failure of the chain, by breakage or by jumping the sprockets, could cause the rope to slip. Severe wear of the chain, bar or sprockets would be dangerous.

7.7 Durability, Reliability, and Maintenance

The critical issues here are related to wear of the chains, sprockets, and pressure bars, and to proper function of the load adapter. Frequent cleaning, lubrication, adjustment, and checking for wear would be needed.

8.0 MOTOR

Currently, Sky Climber uses a single phase AC induction motor with brake (1.3 HP, 1,725 RPM) to drive the Alpha 1000 hoist. As previously shown in Section 4.2, the horsepower rating is consistent with GTRI's calculations for lifting a rated load of 1,000 pounds. In GTRI's calculations, it was assumed that a standard 1,750 RPM AC motor would be utilized. This produced a gear reduction ratio in the range of 120-125:1.

It may be feasible to utilize a 3,600 RPM motor instead of a 1,750 RPM motor with the same horsepower rating. A review of several AC motor manufacturers catalogs has shown that a weight savings of 8-25 percent (4-10 pounds) can be expected if a 3,600 RPM motor is used. This savings would be for a totally enclosed, fan cooled, single phase motor with no brake.

Stepping up to a 3,600 RPM motor would also increase the gear reduction ratio required by a factor of two to a 250:1 ratio. In all likelihood, this will increase the weight of the gear reduction drive. Therefore, it would be extremely important to verify that any increase in the weight of the gear drive would not offset any weight savings realized by utilizing the higher speed motor.

9.0 PRIMARY BRAKE

The primary brake on Sky Climber's Alpha 1000 hoist is part of the drive motor (mounted to the rear). Sky Climber has expressed a desire to move the primary brake from the rear of the motor to the output shaft of the motor. Their belief is that the existing motor could be replaced with a standard off-the-shelf motor, thereby reducing manufacturing costs.

Placement of the primary brake on the rear of the motor or on the motor output shaft is a logical choice. At these locations, the torque is lowest, hence, less braking power is required. For a hoist with a 1,000 pound rated load capacity utilizing a 1,750 RPM motor, nearly 4 ft-lbs. of torque is seen at the motor shaft, while nearly 500 ft-lbs. of torque is seen at the driven drum or sheave (assuming a 9.375-inch diameter).

The desired brake would most likely be a spring-applied/solenoid release brake with a manual override. There are several concerns if the brake is mounted between the motor and the gear reduction drive. First, this configuration does not permit the gear drive pinion (if a planetary gear arrangement is utilized) to be mounted directly to the motor shaft. Second, the housing has to be designed so that the manual release can be accessed.

A review of several brake manufacturer's catalogs has identified a brake that could be utilized. Further examination is needed to determine whether it will be feasible to utilize the brake. There are both weight and cost concerns that must be addressed when selecting the brake configuration. Placement of the brake between the motor and gear drive will most likely increase the design time. This may also be undesirable.

10.0 EVALUATION OF TRACTION MECHANISM CONCEPTS

GTRI presented three traction mechanism concepts to Sky Climber in mid-May. It is believed that each of these concepts met nearly all of Sky Climber's desired capabilities for the next-generation power hoist. The three concepts presented were as follows: dual drum traction hoist, chain traction hoist, and a single-wrap, V-groove traction sheave.

The V-groove traction sheave has not been presented in detail here, because it appears to be similar to Power Climber's Pocket Climber. It is believed that Sky Climber is more familiar with how this type of mechanism operates. The details of the dual drum and chain traction hoist concepts have been presented here to ensure that Sky Climber is familiar with the concepts, and that they will be able to make an educated decision on which concept they would like to use with the next-generation power hoist.

Although it is very early in the design process to compare one design concept to another, GTRI has attempted to compare the various concepts with one another using Sky Climber's desired capabilities as a selection criteria. Two of Sky Climber's goals (target hoist weight of 75-100 lbs. and target manufacturing cost of $1,000) are very difficult to assess at this stage.

10.1 Modularity of Components

Sky Climber desires modularity in regard to both the traction mechanism/gear train, motor, and power supplies. The modularity of hoist parts would need to be incorporated into the design process. This would be feasible for any of the three traction mechanism concepts. The dual drum traction hoist and the V-groove traction sheave would appear to be better than the chain traction hoist at accepting varying wire rope sizes.

10.2 Durability

The dual drum and V-groove sheave concepts would be better suited than the chain traction hoist to operate in a severe environment. There could be serious contamination problems in the chain traction drive that would lead to abrasion wear problems. It would be difficult to cure this problem, as a lubricated wire rope (carrying contaminants) is being brought into a lubricated chain that is susceptible to wear.

10.3 Reliability

The dual drum and V-groove sheave concepts would not be as sensitive as the chain traction drive to varying wire rope sizes. The chain traction hoist might have to utilize different chains for the different rope sizes (one chain for the 5/16-inch and 8 mm diameters and another chain for the 3/8-inch and 9 mm diameters).

The power hoists will be more reliable if they do not allow contaminants to build-up. The dual drum concept would allow for the contaminants to fall out. The V-groove sheave concept would appear to have similar contaminant build-up to Sky Climber's Alpha 1000. There could be considerable contaminant build-up with the chain traction drive concept and associated chain wear problems. As the chain wears, it would also get longer, thus requiring more regular maintenance.

10.4 Ease of Installation

It appears as though all three concepts could be made to accommodate Sky Climber's desired self-reeving feature. Breech loading could also be achieved with all three concepts. The chain traction drive would require that a cover plate be removed and the two chains separated. The dual drum concept would require that a cover plate be removed in order to breech load the hoist. The rope guides between the two drums make it almost impossible to put the wire rope into the wrong groove. To breech load the V-groove sheave concept, the pinch roller(s) would have to be pulled back from the wire rope.

10.5 Ease of Maintenance

The dual drum and chain traction drive concepts would both provide easy access to interior components so that maintenance could be performed. Using Power Climber's Pocket Climber as an example, the V-groove sheave concept might not provide as much access as the other two concepts. At this point in the design process, it is difficult to estimate how many overall components would be required.

10.6 Wire Rope Handling

All three hoist concepts can be designed to accommodate several wire rope sizes. As previously mentioned, the chain traction hoist might have to utilize two different chains to accommodate the four common wire rope diameters.

Of the three concepts, the dual drum traction hoist would probably put the least amount of stress on the wire rope. The chain traction drive could be designed so that a significantly low clamping pressure was exerted on the wire rope. The V-groove traction sheave would probably put the most clamping pressure on the wire rope as the rope is not only being pinched by a roller(s), but it is also being guided to where it exits the hoist.

10.7 Hoist Weight

It is difficult at this stage to determine which hoist concept would be lighter weight and easier to handle. It seems likely that the V-groove sheave concept could be the lighter of the three concepts. Using Power Climber's Pocket Climber as a guide, it seems very probable that the V-groove sheave concept could be designed to meet Sky Climber's 75-100 lb. target weight.

10.8 Manufacturing Costs

It is difficult at this stage in the design process to determine which hoist concept would be cheaper to manufacture. It is also difficult to assess whether any of the three hoist concepts would meet Sky Climber's target manufacturing cost of $1,000.

11.0 RECOMMENDATIONS

Each of the traction mechanism concepts that were presented to Sky Climber in mid-May has merit and could warrant further consideration as part of a new generation power hoist. The two traction mechanism concepts detailed in this report, the dual drum hoist and the chain traction hoist, are both quite innovative when compared to the existing products available in the power hoist industry. This includes both Sky Climber's products and those of their competitors (Power Climber, Tirak, and Hi-Lo).

Although it is not detailed in this report, the single-wrap, V-groove traction sheave is similar to Power Climber's Pocket Climber hoist. It is believed that GTRI could simplify the packaging of the interior hoist components. The method of breech loading would also be simplified.

The chain traction hoist is very different from any of the hoist designs on the market today. The hoist has several advantages: the lower gear ratio reduces the size of the reducer, the hoist treats the wire rope well, and the hoist is easy to self-reeve and breech load. The hoist does have some potential problems: contaminant build-up could lead to severe wear on the chain and sprockets, and failure of the chain could be catastrophic. The hoist might also require more maintenance than the other two hoist concepts.

The dual drum hoist concept treats the wire rope very well. The rope does not rub against itself (like the Model 4) or any stationary object (like the Alpha 1000 and the Pocket Climber). The hoist can be designed to accommodate both self-reeving and breech loading as Sky Climber desires. The low angle helical gearing, used to drive the drums in the same direction, greatly reduces the required torque capacity of the primary reducer.